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NOTES ON ENGINE DESIGN

FURMAN





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NOTES ON ENGINE DESIGN

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P R E F A C E

These notes have to do with an elementary problem in engine design. No specific data are given; the student must choose his own, assuming numerals to represent the horse-power, the revolutions per minute, and the boiler-pressure.

As the practical engineer knows very well, there are certain standard relations between the numerals that may be here chosen. It has been found, however, that although these relations may be told over and over again, that one does not fully realize that there is, for example, a limiting piston speed establishing a relation between length of stroke and revolutions per minute, and establishing also a limit of inertia of moving parts which results in certain relations between horse-power and revolutions per minute, until one has run up against the actual obstruction himself.

In requiring the beginner to assume his own data, he is quite likely to assume an impracticable combination, and it is some advantage for him to do so, for these notes are so arranged that he cannot spend much time before coming face to face with the consequences of a wrong assumption, when another start will give him an experience and an impression that is more likely to be of permanent value than if he had assumed correct data the first time. It is quite possible to make three or four false starts, each of which will shortly lead to a stopping point in the notes and compel a new start. No one can follow through these notes to page 6 without having assumed proper data.

This problem is not intended for a finished exercise in large or complicated multi-cylinder design, but it is complete in the elementary work of engine design so far as the cylinder, valve, steam chest, and stuffing-boxes, etc., are concerned. Time is too limited, except in some special cases, to follow through the detail design of the crosshead, shaft, crank, frame and bearings, etc., but when there is time one of the standard reference books, such as Unwin or Kent, is used for direction. These details of design, are required of all students, although not for this particular problem, in their course in Unwin's Machine Design.

The method of approaching this problem, as here given, is original so far as is known to the writer, and it is because of the very satisfactory results obtained from it in the course of work in the drafting room that these notes are issued in the present form.

FRANKLIN DE R. FURMAN.

• HOBOKEN, N. J., August 18, 1911.

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NOTES ON ENGINE DESIGN.

SECTION I.—ENGINEER'S DESIGN.

1. Steam-engine design, considered as a problem in Machine Design, starts with the diameter of cylinder, length of stroke, revolutions per minute, and initial steam pressure given. Also it must be known, of course, whether the engine is single or double acting and if it is a single, double, triple, or quadruple cylinder engine.

2. The selection of type and determination of the data as mentioned above is the business of the engineer, and depends on the nature of the service and the degree of economy required. The nature of the service will usually determine the rotative speed of the engine shaft, this ranging from very high as in driving steamship propellers and dynamos, through medium speeds, down to very low speed for pump work. The degree of economy obtained depends principally upon the ratio of initial to terminal steam pressure, and the greatest range is obtained through multiple-cylinder engines usually with simple valves and valve-gears; or, where the large horse-power of the multiple-cylinder engines is not required, through a single cylinder engine with a special form of valve or valve-gear, or both, that will give an early cut-off and thus give a large range of expansion.

3. The method of procedure in the design of a steam-engine as a machine is fundamentally the same in all types. Assuming a knowledge of thermodynamics of the steam-engine, and of valves and valve-gears that will give any cut-off to meet desired conditions, we will proceed to lay out the engineer's design of steam-engine of the single cylinder, double-acting plain D-valve type, as permitting the quickest progress, although not giving by any means the greatest economy. The directions, however, will be general so that they could be used for any type of engine were there sufficient time to develop the more economical types.

4. The usual starting point of a steam-engine design is the horse-power and the number of revolutions per minute. If the engine is to be installed in a plant where there is already sufficient boiler capacity to take care of it then that boiler pressure must be also included in the data; or if a boiler is to be installed to run the engine, the engineer has greater freedom in determining the initial cylinder pressure. In this problem, data are to be assumed from the following ranges: Horse-power, 20 to 100; revolutions per minute 150 to 400; and boiler pressure 60 to 100 lbs. above atmosphere.

5. Engines running up to 125 r.p.m. may be termed low speed engines; those from 125 to 250 r.p.m., medium speed; and those above 250 r.p.m., high speed. The ranges here given for low, medium, and high speed are not according to any accepted standard, for, what may be high in one line of industrial work may be low in another. The figures here given are arbitrarily taken, but it is believed they are fairly close to the broad, general idea of low, medium, and high speed engines. Single cylinder high speed engines are regularly listed by several manufacturers, up to 600 r.p.m. but this speed is for small engines only. To give some guide as to practical relation between rotative

engine velocity, length of stroke, and horse-power the following table, made up from catalogue lists of twelve engine manufacturers, is given. The underlying principle compelling manufacturers to limit engines of such high speed as 400 to 600 r.p.m. to small sizes, is based on the stresses induced in the engine by the inertia of the reciprocating parts, which become very large and impracticable for the large masses required for large horse-power:

Table 1.

Revolutions per minute	Stroke in inches.	Bore in inches.	Horse-power.
600	5	5 or 6	11 to 30
500	5 to 6	6 or 7	17 to 43
400	6 to 8	6 to 10	27 to 64
300	7 to 12	7 to 12	41 to 80

Below 300 there are no definitely limited ranges for stroke, bore, and horse-power.

6. It may be that certain desired combinations of data are not practical, and if so the student will run counter with some of these directions as he proceeds with his work. For example, one might select from the ranges given in paragraph (4) a number of revolutions and a horse-power not compatible with accepted engineering practice as shown in paragraph (5). Another example is taken from the following table in which it is shown that there are certain practical limits for the velocity of the piston in feet per minute, for various classes of engines.

Table 2.

Piston speeds.	Feet per minute.
Small stationary engines	300 to 500
Large stationary engines	500 to 1,000
Corliss engines	400 to 750
Locomotives	600 to 1,200

In this problem engines up to 60 horse-power may be considered "small," and those above as "large," engines.

Piston speed should not be confounded with rotative speed. The piston speed of a Corliss engine (which is necessarily a low speed engine) may be greater than the piston speed of a smaller high speed engine, as shown by a 36" stroke Corliss running at 120 r.p.m. with a piston speed of 720 feet per minute against 500 feet per minute for a five-inch stroke engine running at 600 r.p.m. In the former case the engine parts must be comparatively heavy and strong to resist the forces transmitted at each stroke, whereas in the latter case the parts are of comparatively light weight and transmit a smaller force at more frequent intervals.

7. A further general guide in determining data in steam-engine design, is the generally accepted practice of using initial steam pressures between certain limits for various classes of engines as illustrated in Table 3. The underlying principles involved in making up the figures of this Table

is the law of expansion of steam, and its economical use, which requires that its terminal pressure be low. With a single cylinder D-valve engine the terminal pressure cannot be low unless the initial pressure is low. With multiple-cylinder engines the terminal pressure may be low and the initial pressure high even with late cut-offs.

Table 3.

Initial cylinder pressure.	Lbs. per sq. in. (gauge).
Single cylinder engines	60 to 120 lbs.
Compound engines	100 to 180 lbs.
Triple expansion engines	150 to 240 lbs.
Locomotive engines	140 to 200 lbs.

The initial pressure in computing the low pressure cylinders for compound engines may be taken 20 to 40 lbs.

8. The first direct step in determining the proportions of the cylinder may now be taken. In the formula,

$$\text{Horse-power,} = \frac{2 P L a N}{33,000}$$

it is necessary to find the mean effective pressure, length of stroke and the bore of the cylinder. With the boiler pressure assigned and the approximate cut-off determined by the specification of the valve in this problem (see also par. 10), the theoretical indicator card may be constructed and the trial mean effective pressure found.

In constructing the theoretical indicator card it is necessary to make allowances:

(a) In determining initial steam pressure in the cylinder, which is from 6 to 10% lower than boiler pressure, or lower than receiver pressure if a multiple-cylinder engine⁸ is being considered.

(b) For the *steam clearance*, which includes the volume due to *linear clearance* (distance between piston and cylinder head at end of stroke) plus the volume of the steam-port, and is usually taken from 6 to 12% of the volume displaced by the piston. Engines having short, straight ports and a long stroke, such as the Corliss engine, may keep the steam clearance down to ½ to 3%.

(c) For the back pressure, which is usually taken at 1 lb. above atmosphere in a non-condensing engine such as the present problem, or at about 2 inches above condenser pressure if condensing. If a condensing engine is used, allow 26 inches of vacuum in condenser.

(d) For the amount of compression, which usually starts at 85 to 90% of the completed stroke in low speed engines, and at 80 to 85% for high speed engines. The underlying principle here involved in the determination of these figures, is based on a proper balancing by steam pressure of the inertia of the reciprocating parts.

(e) For the perfection of the theoretical card over the approximate practical card which has less area. This falling off is due, principally to condensation, and to the wire drawing produced by the relatively slow closing of the valves, some much more than others; and in engines improperly designed or carelessly maintained, to insufficient port areas, leaky surfaces, etc. "Kent" gives, among others, the following values:

NOTES ON ENGINE DESIGN

Table 4.

Factor for Obtaining Probable Mean Effective Pressure from Theoretical Mean Effective Pressure.

	Factor
Unjacketed cylinder, ordinary valve and gear	0.80 to 0.85
Jacketed " " " " " "	0.90 to 0.92
" " special " " " "	0.94

9. All the data are now available for the construction of the theoretical indicator card as shown in the accompanying diagram (Fig. 1) from which the theoretical mean effective pressure may be obtained graphically by taking the average of the lengths of the fine vertical lines shown at m , n , o , p ,

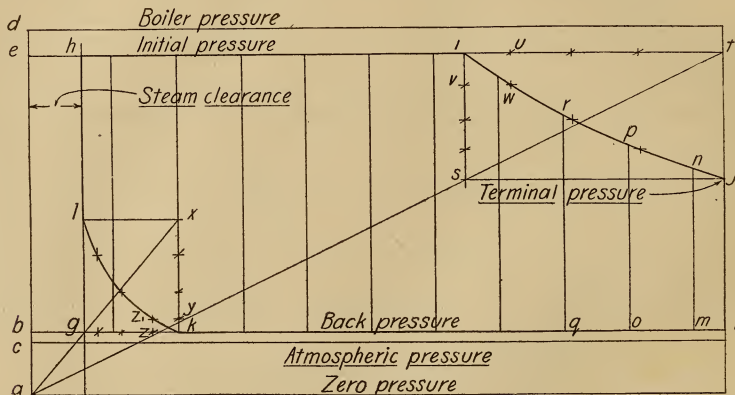


FIG 1.

q r , etc., on the diagram, or by other graphical methods, or by computation as given in "Kent," or by the planimeter. From this the probable mean effective pressure may be obtained and used in the formula given in paragraph (8).

10. In the present problem the trial cut-off for the plain D-valve may be assumed at .55 to .6 stroke; and in computing the port width the live steam velocity may be taken at 8,000 to 10,000 ft. per minute, and the exhaust at 6,000 to 8,000 ft. per minute. Trial cut-off is stated because it is desirable to keep the cylinder dimensions in round numbers when finally found; and as they will probably work out in quarters, eighths, or sixteenths of an inch when first determined, they can be most readily changed to whole numbers (or halves of an inch are often used in small work) by manipulating the cut-off and thus altering the mean effective pressure, while still retaining the exact number of revolutions required. The final cut-off should fall between .5 and .7. The live and exhaust steam velocities are taken here rather high for the assigned range of steam pressures, but by so doing the valve travel will be kept down with the plain D-valve.

11. Had the boiler pressure not been given in this problem the construction of the expansion curve could have been started at the point *j*, Fig. 2, at any desired terminal pressure; then the expan-

sion line would be carried backward until it had reached the vertical through the desired cut-off point, and its elevation at that instant would determine the initial steam pressure. For example should 20 lbs. above atmosphere be the desired terminal pressure of the steam, the initial pressure for a valve giving .7 cut-off would be only 33 lbs.; very little of the expansive power of the steam would be made use of, and the mean effective pressure would be low, and an engine having unnecessarily large bore and stroke would be required for a given horse-power. Whereas with a valve cutting off at .2 stroke it will be seen that the initial pressure is large (115 lbs. gauge in Fig. 2), that there is a wide range of expansion of the steam, and that the mean effective pressure will be high, thus giving an economical engine with comparatively small bore and stroke for a given horse-power.

In multiple-cylinder engine specifications the engineer may, in a broad, general way, apportion the total load equally among each of the cylinders; and the bore and stroke of each may then be

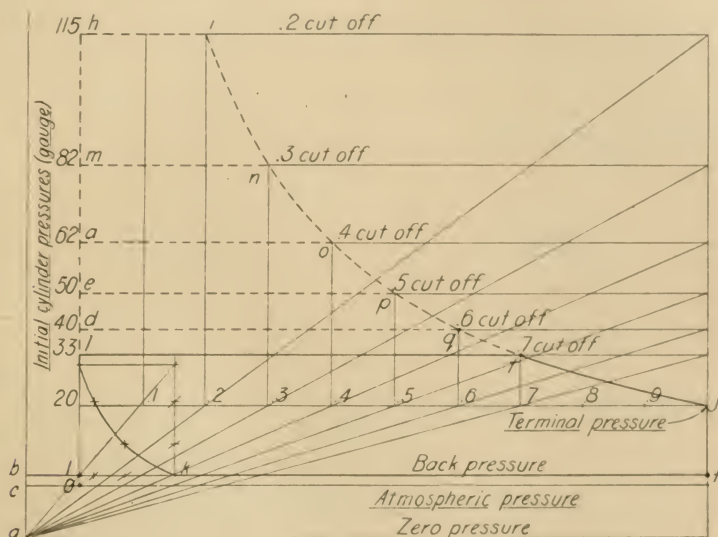


FIG. 2.

determined by working backward from an assumed final terminal pressure, the initial pressure of the low cylinder being the terminal of the intermediate after making due allowance for receiver loss between the cylinders.

12. After finding the probable mean effective pressure, as in paragraph (9), all the data in h.p. $= \frac{2 P L a N}{33,000}$ are known excepting L and a . Since $2 L N$ = piston speed for which there are generally accepted values for various classes of engines, as given in Table 2, paragraph (6), L may now be found, when the above formula may be solved for a and the diameter or bore determined. The ratio of $\frac{L}{d}$ varies from about 85 to 2, the lower values being used for the highest speed, with the average value at about 1.2 for ordinary cases. This completes the engineer's design, providing the

bore and stroke come out in even numbers. Sometimes half inches are permitted on the bore but rarely on the stroke and the student will observe this in his final specifications.

Should the length of stroke not come out in even inches take the next larger or next smaller whole number according to judgment and work back to a new piston speed which should fall within the limits of Table 2. With the new piston speed, solve for a again and find d , the diameter of the cylinder. Should it come out in anything but whole or half inches take the next larger or next smaller half inch, if below 9"; or the next larger or next smaller whole inch, if above 9", according to judgment. Since the bore and stroke are now reduced to round numbers which must be retained, and since the horse-power and revolutions must remain as originally assigned, there is only one variable left and that is the mean effective pressure and its new and final value may be found by solving for P , where the revised values for L and a and the original values for N and h.p. are used. According as the revised mean effective pressure comes lower or higher, the expansion line ij of Fig. 1 may be lowered or raised by the method of trial and error and the necessary point of cut-off thus determined to give the required horse-power and speed. This final cut-off is the one to be used in laying out the Zeuner diagram in designing the valve.

From the figures now obtained fill out the spaces in the statement of the following problem and proceed with the machine design.

SECTION II.—DRAFTSMAN'S DESIGN.

Problem: Design details for a steam-engine cylinder having a diameter of _____ inches, stroke of _____ inches, initial steam pressure _____ lbs., revolutions per minute _____.

13. Design to be made on full-size sheet (D. E.). Arrangements as shown in Fig. 3. Use the largest available scale.

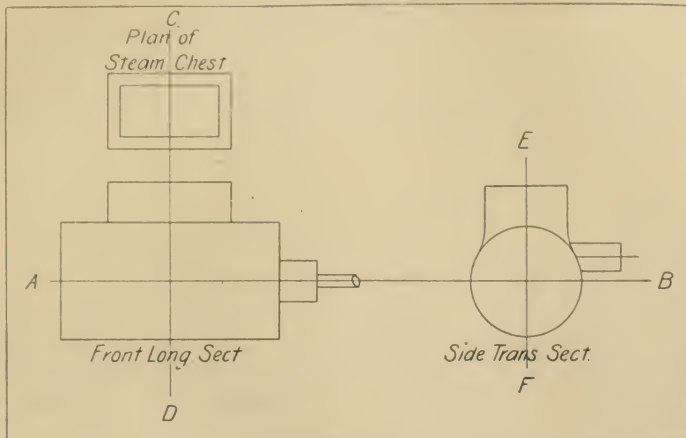


FIG. 3.

14. These notes are intended merely as suggestions in directing the work in the Drafting Room. Fuller descriptions may be found in the following books of reference:

"A Manual of Marine Engineering," by A. E. Seaton.

"A Manual of Machine Drawing and Design," by Low & Bevis.

"Easy Lessons in Mechanical Drawing and Machine Design," by J. G. A. Meyer.

"Machine Design," by Professor Unwin.

"Steam-Engine Design," by Prof. J. M. Whitman.

"The Steam-Engine," by Prof. Wm. D. Marks.

"Seaton & Rounthwaite's Pocket-Book of Marine Engineering."

"Spooner's Machine Design, Construction and Drawing."

"Kent's Mechanical Engineer's Pocket-Book." (This book must be kept at hand throughout the design for ready reference.)

15. The formulæ given in the several text-books for use in computation in designing frequently give impracticable results. A single formula for a certain detail cannot be expected to apply to all sizes and conditions of service. Where more than one formula is given in these notes for proportioning any one part, *solve by all formulæ and then determine the figure to be used according to your own judgment.* Where calculated results conflict with practical observations based on successful practice, neglect the calculated results, and use the proportions that are known to have given satisfactory service.

16. In the present problem the center-lines AB , CD , and EF of Fig. 3 are to be drawn only after the over-all dimensions have been determined by summing up the preliminary calculations from which a free-hand scale drawing must be made *embodying only the broader essential features of each of the parts*. This free-hand sketch is to be made on the computation sheets using the squares of the cross-section lines to determine the scale. This sketch, which is made from calculated values, should be critically examined to see if it *looks reasonable* in one's own common-sense judgment. If it does not, the computation should be gone over again, and if it still comes out the same, the person in general charge of the work should be consulted. Do not go ahead against your own judgment until you have to and then first fortify yourself with some authority.

Sometimes it is not necessary or advisable to work out *all* parts in preliminary sketches before proceeding to lay down the center-lines on the design sheet. Some things may be assumed on the basis of previous experience, and the greater one's experience in the work in hand the quicker he may plan the arrangement of views. In the student's case, for example, he could quickly figure port

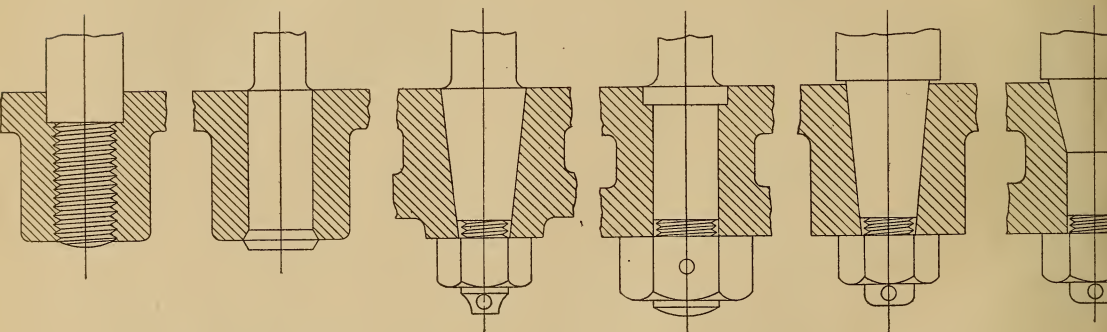


FIG. 4.

FIG. 5.

FIG. 6.

FIG. 7.

FIG. 8.

FIG. 9.

width, and then in sequence assume approximately the bridges and exhaust port, height of valve-seat above cylinder, height of valve and height of steam-chest, and so determine nearly enough to lay out center-lines, the distance from the center-line of the front view to the top of the front view, without waiting to work out all these details with precision. In the larger engines there may not be room to put in the full top view, or steam-chest plan, in which case one-half may be shown.

17. It is to be noted at the outset that the illustrations given in these notes are intended to be suggestive rather than exact patterns, and therefore in some cases all the lines necessary for completeness of projection are not shown. The missing lines should be provided by the student on his own work.

18. In order to properly shape the cylinder head, as it will presently appear, the first step in the design will be to lay out the piston and piston-rod in place at the head end of the stroke.

19. Sizes and forms for piston-rods and piston-rod ends, from which the student may choose, are here shown. Piston-rods are usually made of steel, sometimes of hammered wrought iron.

20. Area of piston-rod at root of thread = $a = \frac{1}{4} \pi \frac{D^2 P}{f}$ to resist tensile strain of steam pressure.

a = area of rod at root of thread. $f = 5000$ for wrought iron,
 D = diameter of piston in inches. $= 7000$ for steel.
 P = steam pressure per square inch.

There is also authority for $d_1 = \frac{D}{80} \sqrt{P}$ for wrought iron,

$$d_1 = \frac{D}{89} \sqrt{P} \text{ for steel.}$$

d_1 = diameter of rod at root of thread.

These latter formulæ are deduced from the expression:

$$\frac{1}{4} \pi d_1^2 f = \frac{1}{4} \pi D^2 P, \text{ from which } d_1 = D \sqrt{\frac{P}{f}}.$$

f is here taken at $= 6400$ for wrought iron. Hence $d_1 = \frac{D}{80} \sqrt{P}$.

21. Size of body of rod $= d = K D \sqrt{P}$.

For short-stroke engines, the value of K ranges from 0.0169 to 0.0182.

Another authority gives $d = \frac{D}{60} \sqrt{P}$ for wrought iron.

$$d = \frac{D}{69} \sqrt{P} \text{ for steel.}$$

22. For long-stroke engines with piston-rods whose lengths $= 20 \times$ diameter, or more, the following formulæ may be used:

$$d = 0.04 \sqrt[4]{D^2 l P} \text{ for wrought iron,}$$

$$d = 0.038 \sqrt[4]{D^2 l P} \text{ for steel.} \quad l = \text{length of rod.}$$

$$\text{Another safe formula is } d = \frac{D}{48} \sqrt{P}.$$

By taking $f = 3600$ for wrought iron and 4800 for steel in the formulæ in paragraph 20, due allowance is made for bending tendency of long piston-rods.

23. In practice some engine builders have found from experience that a rod $\frac{1}{6}$ to $\frac{1}{7}$ of the diameter of the cylinder gives satisfactory results, and it is often made so without any calculation.

24. The end of the piston-rod must be a perfect fit. Rods are usually tapered, or made conical at the end to permit easy withdrawal. With piston end having a large taper, a shoulder should be used to prevent drawing in cone and splitting the piston, or piston hub must be made extra heavy. See Figs. 6, 8 and 9.

25. When the piston taper runs the whole depth of the piston it may be at the rate of $\frac{1}{2}$ to $\frac{3}{4}$ of an inch per foot. Even with this taper the piston-rod is sometimes removed with difficulty and, to overcome this the piston shown in Fig. 9 is often used.

26. The shoulder in Fig. 9 is $\frac{1}{16}$ " for small, and $\frac{1}{8}$ " for large engines. Taper 3" to 1 foot on the diameters until diameter of outside of thread is reached; then turn the remainder parallel. The rod should fit with $\frac{1}{16}$ " between the piston and shoulder for small engines, and $\frac{1}{8}$ " for large. Should the diameter of the piston-rod outside of the piston turn out from the calculations, paragraph 21, to be smaller than the shoulder diameter in Fig. 9, the shoulder may be imbedded in the hub of the piston, as in Fig. 7, and the body of the rod turned down.

27. Rods with collar forged on, as in Fig. 7, are often used in heavy work. The conditions shown in Figs. 4 and 5 are used principally in very small and inexpensive work. Fig. 5 has cylindrical end, with piston shrunk on and end riveted.

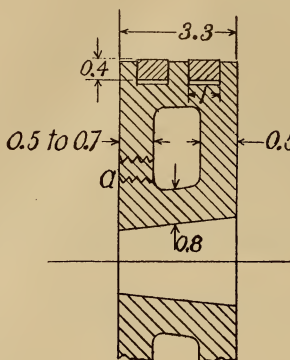


FIG. 10.

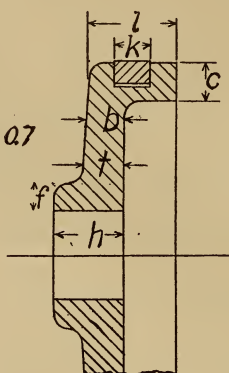


FIG. 11.

28. Pistons are made in a great variety of forms, principally of cast iron and steel. Those shown in Figs. 10 and 11 are the most common forms for small engines.

29. In Fig. 10, the dimensions shown are in terms of the unit, $t = 0.0083 D \sqrt{P}$ for cast iron.

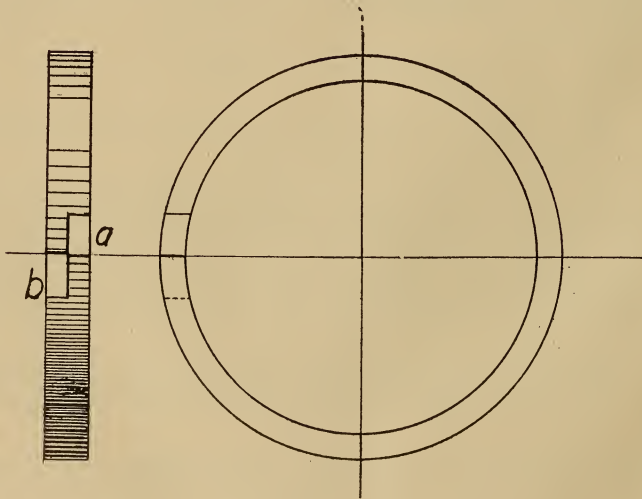


FIG. 12.

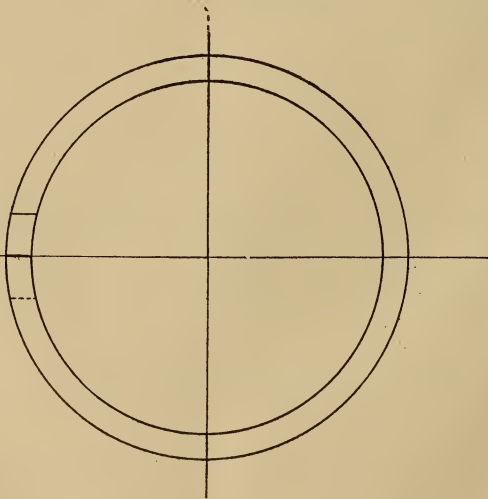


FIG. 13.

(The "unit method" of computing sizes is common in empirical design work especially in cases where exact knowledge of all stresses involved is not known. In this paragraph the unit is arbitrarily taken as t and the thickness of metal in Fig. 10 is based on t in the ratios represented by the

numbers there shown.) D = diameter of cylinder, and P = initial steam pressure. This type of piston is strengthened by thin webs connecting the two walls. Number of webs, or ribs, = $0.08(D + 34)$. It must also be tapped at a to remove the core, and a core-plug inserted. The width of piston should check fairly well with the result given by $\sqrt[4]{Dl}$ when D is large. l = length of stroke in inches.

30. In Fig. 11, $t = 0.0067$ to $0.008 D \sqrt{P}$; $b = 0.86 t$; $c = t$; $f = 0.95 t$; $h = 2$ to $3 t$. The value of k varies widely. It may be taken from one-half to whole diameter of piston-rod, when one ring is used; l = width of packing-ring + $\frac{1}{2}$ to $\frac{3}{4}$ ". Depth of single packing-ring may be 50 to 75% of width. Also k is in practice frequently made $\frac{3}{4}$ " for diameters of 16" and over, and $\frac{1}{2}$ " for smaller diameters. This type of piston is made with wall from t to b anywhere from vertical to a slant of 1 to 3. This form of cast-iron piston should not be used for cylinders of more than 9 inches bore.

31. In making the packing-ring (Figs. 12 and 13), a solid ring $\frac{1}{8}$ " greater than the diameter of the cylinder is turned. Notches a and b are cut halfway through, and the lengths made equal to $3\frac{1}{2} \times (\text{diameter ring} - \text{diameter cylinder})$. The edges are then pressed together, and a pin driven through to hold it in position, while the ring is again turned to the diameter of the cylinder. The pin is then removed, the ring sprung in place over the piston, and both placed in the cylinder. Packing-rings are also often split by a diagonal cut across the ring, instead of the box cut shown in Fig. 12.

32. With the ordinary sprung ring $\frac{1}{8}$ " may be allowed for space between the packing-ring and the bottom of groove of the piston. Steam is sometimes admitted behind the ring to secure pressure on the cylinder-wall, but has not proved very satisfactory. It has been found that a pressure of the ring of 3 to 4 lbs. against the cylinder-wall will prevent leakage against a pressure of 100 lbs.

33. In some designs, especially for small quick-acting pumps, a series of grooves, without packing-rings, on a closely turned piston, serve fairly well to prevent leakage, by reducing the energy of the motion of the steam as it enters each successive groove. This class of piston is made rather wide—not less than $\frac{3}{4}$ the diameter of the cylinder. (See Fig. 14.)

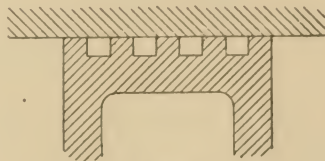


FIG. 14.

34. In the pistons so far mentioned the packing-rings must be sprung in place. In larger engines and in more important work the pistons are built up so as to insert and remove rings without springing, and also to permit of the use of various forms and combinations of packing-rings.

35. Fig. 15 shows a cast-steel piston, the proportions of which are shown. $t = D/200 \sqrt{P}$. $b = 0.7 t$, $f = t$, $h = 4$ to $5 t$. Slope = 1 to 3. A follower-plate, or junk-ring, q , is bolted in place after the packing-rings, m and n , and the bull-ring, o , are adjusted. The follower-plate bolts, which are of wrought iron or steel, are secured in gun-metal nuts, shown at r , so as to prevent rusting. The ring, o , is a split ring, and springs are placed in the space a .

36. Figs. 16 and 17 show two views of a "built-up" cast-iron piston that has given excellent service. The casting, a , is the "spider." b is the "following-plate," o the "bull-ring." For large cylinders the number of ribs for such a piston = $\frac{D}{10} + 2$, and the thickness of each = $\frac{D}{300} \sqrt{P}$. Diameter of follower-bolts = $\frac{D}{400} \sqrt{P} + \frac{1}{4}$ ".

37. In pistons of the built-up type the pressure springs (shown at *E*, Fig. 17) are spaced all around the piston in vertical engines; in horizontal engines those at the bottom are sometimes replaced by solid blocks to support the weight of the piston. In designing the piston, keep the weight a minimum within the bounds of safety.

38. In selecting a piston type for *vertical* engines avoid one whose boss, or hub, would extend toward the *under* side, and require coring the crank-end cylinder-head around the piston-rod to accommodate the extended piston-boss. Such a core would fill with water and cause the engine "pound."

39. With the form of the piston known, the thickness of the cylinder-wall may be determined and the cylinder-heads designed.

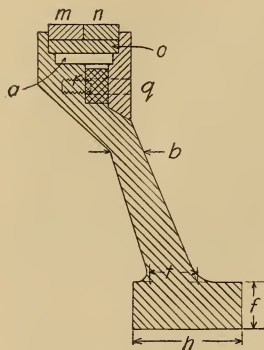


FIG. 15.

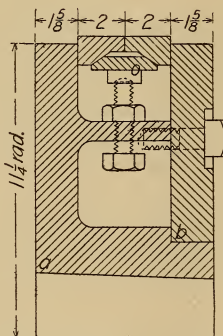


FIG. 16.

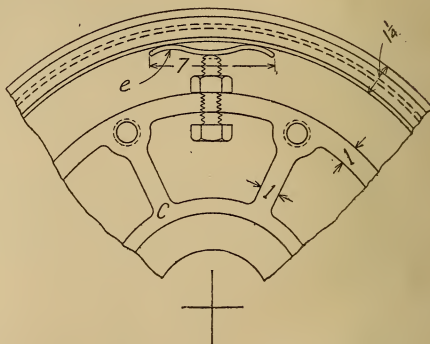


FIG. 17.

40. The thickness of the cylinder-wall must be sufficient:

- (a) To resist internal steam pressure.
- (b) To insure a sound casting.
- (c) To resist strains in handling (in large engines particularly).
- (d) To allow for reboring once or twice when worn.

A thickness of $t = \frac{\frac{1}{2} D P}{S}$, where $S = 3,000$, would resist the internal steam pressure. A thickness of at least $\frac{7}{16}$ " should be used for any cylinder with a bore of 6" or over.

Items (c) and (d) cannot be determined in advance.

To cover all the above cases numerous empirical formulæ, based on practical observations, have been devised. The following may be used as a guide in determining the thickness of the cylinder-wall:

$$t = 0.0001 P D + 0.15 \sqrt{D}.$$

$$t = 0.0003 P D + 0.375''.$$

$$t = \frac{P D}{2,500} \text{ to } \frac{P D}{3,500} + \frac{1}{2}''.$$

41. The cylinder should be counterbored at each end to prevent packing-rings wearing a shoulder at the end of the stroke, and also to allow for reboring. The *diameter* of the counterbore

may be $\frac{1}{8}$ " greater than the cylinder diameter in small engines, and $\frac{3}{16}$ " for larger engines. The packing-ring should overtravel the counterbore about $\frac{1}{8}$ its width.

42. After drawing cross-section of piston in place at head end of stroke, draw the counterbore shown at *a*, Fig. 18. Next calculate width of steam-port, considering the velocity of the exhaust steam as 8,000 feet per minute for engines with a bore of more than 9 inches, and 6,000 feet per minute for 9-inch bore or less. Make the length of port $\frac{3}{4}$ diameter of cylinder. The point *c* of

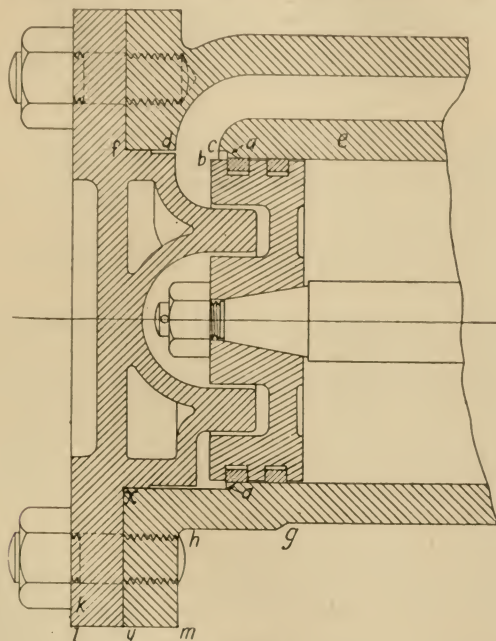


FIG. 18.

the port may be taken directly over *b*, or a small distance to the right. cd = width of steam-port + $\frac{1}{16}$ " to allow for friction of steam on the rough-cast port surface. Make the thickness of wall *e* equal to cylinder-wall.

43. Lay off $df = 1.5 \times$ thickness of cylinder-wall. *f* then lies in plane of cylinder-flange surface. This is generally made flat the whole width of the flange, as shown, and some form of gasket used to make the joint steam-tight. Another plan is to make the flange surface about $\frac{3}{8}$ " wide radially from *f* and make a steam-tight joint without gasket by scraping. See illustration in Fig. 22. The value $\frac{3}{8}$ " is taken because flaws in castings, which might happen to come on such a faced surface, are not, as a rule, more than $\frac{3}{8}$ ". Beyond the scraped joint the rough-cast surfaces of the flanges may each be set back $\frac{1}{32}$ " on small engines, and $\frac{1}{16}$ " on larger ones.

44. The thickness of the cylinder-wall flange may be 1.25 to $1.5 \times$ thickness of cylinder wall. The metal taken at *a* for the counterbore must be replaced at *g*, the distance *ag* being made conventionally equal to the diagonal of a square having the thickness of the cylinder wall for its sides.

45. The width of the cylinder-flange, $x y$, Fig. 18, depends on the size and kind of bolts used. Through bolts require a wide flange to make room for head, or nut, under the cylinder-wall; they also cut into the "lagging" (wood or metal covering usually placed on cylinder between flanges). Felt, asbestos, or other non-conducting substance, is placed between lagging and bare cylinder-wall, to prevent undue radiation of heat. Stud-bolts are generally used. The size and number of these should be so chosen:

(a) that a working stress of not more than $[(\text{area at bottom of thread})^{\frac{5}{12}} \times C]$ per square inch falls on them. ($C = 5,000$ for wrought iron or mild steel);

(b) that the distance between their centers (measured on the bolt circle) should be about 5 or 6 diameters apart, in order to hold the cylinder-cover flange firmly to its seat and prevent escape of steam between the bolts.

Cylinder-bolts less than $\frac{3}{8}$ " in diameter should not be used, except on very small work, on account of liability to excessive strain by wrench in screwing up the nut.

A shop rule sometimes used, is to make the diameter of the bolt = $\frac{1}{2}$ (thickness of cylinder flange \times thickness of cylinder-cover flange). The cylinder-cover flange may be made 1.15 to 1.25 times the thickness of cylinder-wall.

Some authorities give one bolt for each inch of diameter, but this gives a rather larger number than is found in the latest practice.

46. The tap for stud-bolt in cylinder-flange may be made *about* flush with the cylinder-wall, as shown at h , Fig. 18. With the position and diameter of bolt thus shown, the radius of the bolt circle is determined. With a standard nut on the bolt, and an allowance of $\frac{1}{4}$ " at $k l$, the minimum width of cylinder-flange is known. The cylinder-flange width is often made 3 bolt diameters.

47. The cylinder-head flange, $y l$, Fig. 18, has a thickness of about 1.15 to 1.25 \times cylinder-wall, and should always be less than that of the cylinder-flange $y m$, for, in case of accident, the latter is the more difficult and costly to replace. In some shops the stud-bolts are weakened by a groove so that the bolts will break instead of the flanges in case of accident.

48. The inside line of the cylinder-head must conform with the outline of the piston as closely as possible, to keep down the *engine clearance*, which is the *volume* inclosed between the piston and the cylinder-cover at the end of the stroke + the *volume* of the ports.

49. The *piston clearance* is a *linear distance* between the piston and the cylinder-head at the end of the stroke, and is necessary to allow for wear in the two connecting-rod ends and in piston-rod end; also for roughness of casting of the cylinder-head and piston-walls. From $\frac{1}{16}$ to $\frac{1}{8}$ " is sometimes allowed for the latter, and $\frac{1}{16}$ " for each of the three working joints in horizontal engines of the sizes in this problem. The following is a reliable table for piston clearance in vertical engines:

Piston Clearances for Vertical Engines (for horizontal engines use head-end clearance for both ends)

Diameter of cylinder.	Head end.	Crank end.
Up to 14".....	$\frac{1}{4}$ "	$\frac{3}{8}$ "
15" " 20".....	$\frac{3}{8}$ "	$\frac{1}{2}$ "
21" " 40".....	$\frac{1}{2}$ "	$\frac{5}{8}$ "
41" " 60".....	$\frac{5}{8}$ "	$\frac{3}{4}$ "
61" " 80".....	$\frac{3}{4}$ "	$\frac{7}{8}$ "
81" " 100".....	$\frac{11}{16}$ "	$\frac{15}{16}$ "
above 100".....	$\frac{3}{4}$ "	1"

50. The thickness of the cylinder-cover between the flanges may be made = to that of the cylinder-wall for smaller engines. Covers for medium-sized engines may have the same thickness strengthened by radial ribs on the outside, or they may be cast hollow, as shown in Fig. 18, if the

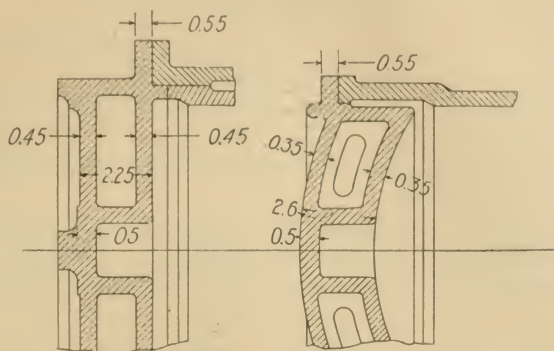


FIG. 19.

FIG. 20.

piston form permits. Covers for large cylinders are always made hollow with internal stiffening ribs.

51. Figs. 19 and 20 show cylinder-heads with principal proportions for large cylinders. The figures given are in terms of the unit $\frac{DVP}{100}$.

52. The cylinder-cover for the crank end may next be constructed. Here, also, the inside

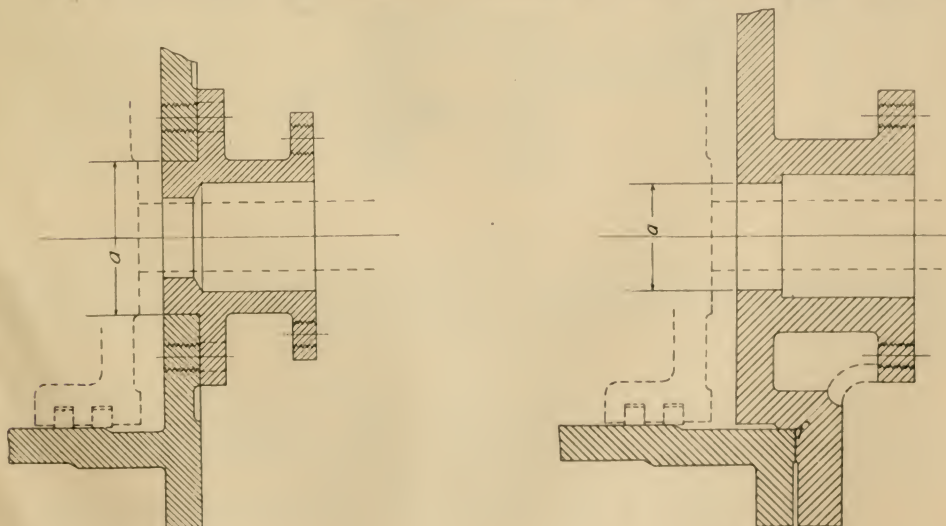


FIG. 21.

FIG. 22.

line should follow the piston contour at a distance equal the piston clearance. In small engines the crank-end cover may be cast solid with the cylinder-wall, as shown in Fig. 21. When this is done the opening, a , should be large enough to admit the boring-bar in turning or truing the cylinder. When the stuffing-box is a part of the removable cylinder-cover, as in Fig. 22, the size of the opening at a for the purpose of admitting a boring-bar need not be considered.

53. With the cylinder-wall and cover cast in one piece, as in Fig. 21, the flange is added just the same. It is necessary for bolting the cylinder to the engine-frame.

54. The cylinder-cover shown in full lines in Fig. 22 may be made as a hollow casting by adding the wall shown in dotted lines.

55. The piston-rod stuffing-box may now be designed. In Figs. 23 and 24, S is the stuffing-box, G the gland, and B, B, B are bushings.

56. Figs. 23 and 25 show two distinct methods of detail for stuffing-box construction, only one of which the student should adopt. The piston-rod is not allowed to come in contact with the cast iron on account of rust forming when the engine is still any length of time, and injuring the rod.

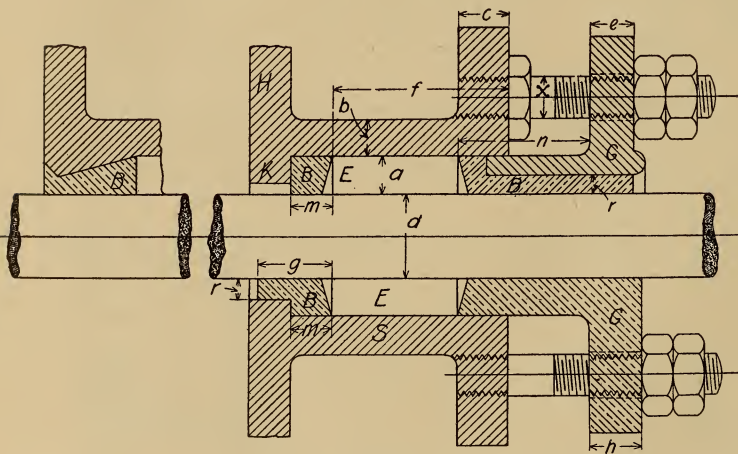


FIG. 25.

FIG. 23.

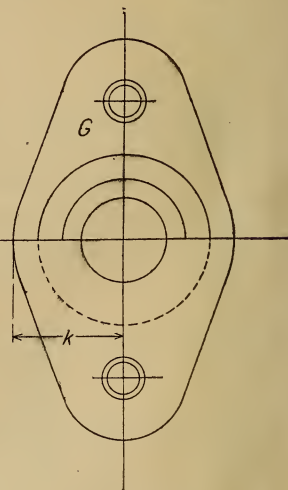


FIG. 24.

The solid "brass," or gun-metal gland in the lower-half design of Fig. 23 is used on smaller engines, and on increasing sizes up to the point where the cost of necessary brass is less than the cost of the smaller amount of the same metal used in the upper-half bushing plus the cost of the iron gland and the machining of the same.

57. For low-pressure engines the space E around the piston-rod is filled with greased hemp rope, asbestos, or various other brands of compressible packing. Numerous patent metallic packings are on the market, and are coming into general use, especially for high-pressure engines. Hemp and similar packings are not serviceable under high temperatures.

58. The small bushing B in the upper half prevents the packing from entering the cylinder. It is not continued through the cylinder-cover, as is the bushing in the lower half. The latter is

largely used, although some makers object to it on the ground that the small neck is apt to break off and fall into the cylinder, thus damaging the cylinder-wall and piston. This may be prevented by the construction shown in Fig. 25, the tapered bushing being made a little too long at first, and then riveted over into the countersink formed in the cylinder-cover. The thickness of the cylinder cover at K ranges from the value at H in small engines to the value of b for larger work. The depth of bushing at g may be taken about $2m$.

59. Definite rules cannot be given for stuffing-box proportions that will suit under all circumstances. Frequently the designer makes the packing space to suit the size of a special brand of metallic packing that has proved satisfactory. The following tabulated data, and formulæ given in paragraphs 60 to 63 for the more important dimensions, represent good practice:

Diameter of rod.	Width of packing space.	Depth of packing.	Depth of neck ring.	Thickness of bushing.	Number and diameter of bolts.	
d	a	f	g	r	No.	x
$\frac{3}{4}"$	$\frac{3}{8}"$	$1\frac{1}{8}"$	$\frac{3}{4}"$	$\frac{3}{16}"$	2	$\frac{3}{8}"$
1"	$\frac{5}{16}"$	2"	$\frac{5}{8}"$	$\frac{3}{16}"$	2	$\frac{1}{2}"$
$1\frac{1}{4}"$	$\frac{7}{16}"$	$2\frac{1}{4}"$	$\frac{7}{8}"$	$\frac{1}{4}"$	2	$\frac{1}{2}"$
$1\frac{1}{2}"$	$\frac{1}{2}"$	$2\frac{1}{2}"$	1"	$\frac{1}{4}"$	2	$\frac{5}{8}"$
$1\frac{3}{4}"$	$\frac{1}{2}"$	$2\frac{3}{4}"$	$1\frac{1}{8}"$	$\frac{1}{4}"$	2	$\frac{3}{4}"$
2"	$\frac{9}{16}"$	$3\frac{1}{8}"$	$1\frac{1}{4}"$	$\frac{1}{4}"$	2	$\frac{3}{4}"$
$2\frac{1}{4}"$	$\frac{9}{16}"$	$3\frac{1}{2}"$	$1\frac{3}{8}"$	$\frac{5}{16}"$	2	$\frac{7}{8}"$
$2\frac{1}{2}"$	$\frac{5}{8}"$	$3\frac{3}{4}"$	$1\frac{1}{2}"$	$\frac{5}{16}"$	2 3	$\frac{7}{8}"$ $\frac{3}{4}"$
$2\frac{3}{4}"$	$\frac{5}{8}"$	4"	$1\frac{5}{8}"$	$\frac{5}{16}"$	2 3	1 $\frac{7}{8}"$
3"	$\frac{11}{16}"$	$4\frac{1}{4}"$	$1\frac{3}{4}"$	$\frac{5}{16}"$	2 3	1 $\frac{7}{8}"$
$3\frac{1}{4}"$	$\frac{11}{16}"$	$4\frac{5}{8}"$	$1\frac{7}{8}"$	$\frac{3}{8}"$	2 3	$1\frac{1}{8}"$ $\frac{7}{8}"$
$3\frac{1}{2}"$	$\frac{3}{4}"$	5"	2"	$\frac{3}{8}"$	2 3	$1\frac{1}{8}"$ 1

In the table on this page the values of a are based on a solid gun-metal gland. With cast-iron glands bushed with gun-metal a must be increased $\frac{1}{8}"$ in each case. For hemp packing, and piston-rods under $1\frac{1}{2}"$ diameter the following rule for a may be used:

$a = \frac{1}{4}d + \frac{1}{16}"$ or $\frac{1}{8}"$. Take m equal to a , or a little less.

60. Fig. 23, $b = \frac{1}{16}d + \frac{3}{8}"$ (minimum), $c = 1\frac{1}{4}$ to $1\frac{1}{2}x$, or $\frac{5}{4}(\frac{1}{4}d + \frac{3}{8}"$). A rule for the diameter of the stud, x , is $\frac{1}{8}d + \frac{1}{2}"$. Draw the bolt, x , with the threads flush with the stuffing-box wall, thus permitting the flange to be made narrower. The studs are often forged with hexagonal collars, as shown, for convenience in screwing in place, and for bearing surface when jamming the thread. A circular collar is sometimes used for the latter purpose. Frequently these studs are made without collars, to save expense, in which case the seating of the nut depends on the jamming of the end thread.

61. The diameter, or length, of the stuffing-box flange may be such that the line at c falls a distance $x + \frac{1}{8}''$ to $\frac{1}{4}''$ from the center-line of stud. Length of gland $n = \frac{5}{8}$ to $\frac{4}{5}f$. The distance the gland may be moved in should be $\frac{1}{2}$ to $\frac{2}{3}f$ so that the packing may be compressed or taken up by this amount if desired. Thickness of gland-flange, e , when of brass or composition metal should be equal to c , as shown at h ; when of iron, it should be less than c , $1\frac{1}{4}x$ or $\frac{1}{4}d + \frac{3}{8}''$ giving fair values. The lock-nuts should have a combined depth of $1\frac{1}{2}x$. In practice they are used in two ways: 1st, one nut is of standard depth, and the other nut $\frac{1}{2}$ standard depth; 2d, both nuts are of the same depth $= \frac{3}{4}x$. The last method is preferable. In every case the nuts are of standard diameter, to permit use of standard wrenches or spanners.

62. The end view of the stuffing-box and gland, Fig. 24, shows a common form of the same. $k = \frac{1}{2}d + a + b + \frac{1}{8}''$ to $\frac{1}{4}''$. If the form of the stuffing-box gland is made circular, as it frequently is, it may receive a finish by turning at the same time the cylinder-flange is turned, with very little extra expense. The surface for the form shown in Fig. 24 must remain rough cast, or have a more expensive hand finish. The gland-flange may be made a trifle smaller than the stuffing-box flange, as shown. Very often the stuffing-box flange and gland-flange are made circular in which case three stud-bolts are generally used, each bolt having a diameter of about $x_1 = \frac{1.6}{\sqrt{N}} \times (0.12d + 0.4'')$. N = number of bolts used. (See Fig. 27.)

63. Fig. 26 shows one-half of a stuffing-box using a popular metallic packing consisting of metal rings. The small spaces form annular recesses for collection of water due to condensation. A

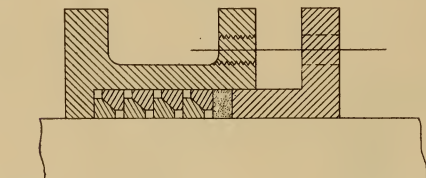


FIG. 26.

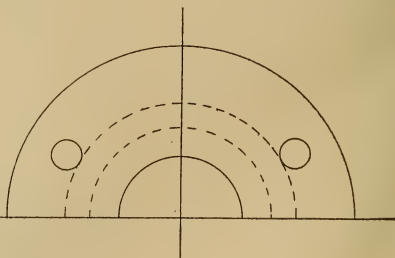


FIG. 27.

fibrous ring is placed between the gland and packing to facilitate lubrication. With this style of packing the value n in Fig. 23 may be considerably less, as there is comparatively little compression of the packing.

64. After designing the stuffing-box, the height of the valve-seat may be determined. This should be as close to the cylinder-wall as possible, in order to save both space and weight in the cylinder casting. Its minimum elevation is governed by the necessary sectional area in the exhaust passageway at x , Fig. 29, leading to the exhaust-pipe B . Sometimes the position of the eccentric on the shaft (which should be in line with the valve-stem), requires that the elevation of the valve-seat should be greater than that called for in the previous sentence: in this design the latter point need not be considered. Before determining the minimum value of the cross-section, x , the width of the exhaust-port, a , b , Fig. 28, should be known. This distance depends upon the exhaust-lap of the valve, which must therefore be laid out at this point of the design.

65. Construct a plain D slide-valve that will give the theoretical indicator card already drawn, paragraph 8. Take lead = $\frac{1}{32}$ " on engines of 9" diameter and under, and = $\frac{1}{16}$ " on all over. In this exercise both ends of the valve may be made symmetrical. Should the exhaust-lap come out negative, change it arbitrarily in this problem to zero and correct the indicator card accordingly. This change will ordinarily not effect the mean effective pressure beyond the range of allowances already made. Assume that connecting-rod equals five crank lengths.

66. Lay down the valve-seat, $a b$, at a trial elevation and draw a trial contour for the exhaust port, $a g h b$, thus determining a trial average width, y . This work should be lightly drawn, as the following consideration may make a change necessary.

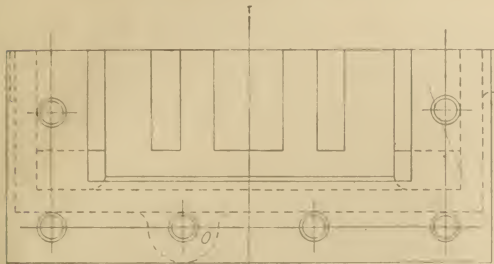


Fig. 30

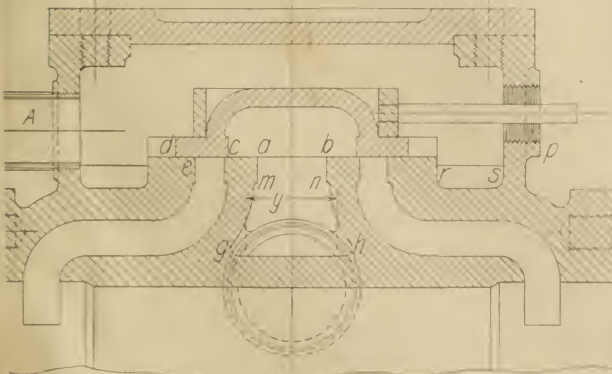


Fig. 28.

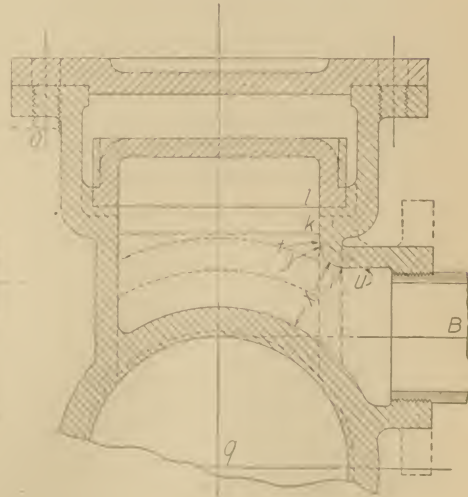


Fig. 29.

67. In Fig. 29, draw the center-line of the exhaust-pipe, B , so that when produced it is tangent to the inside of the cylinder-wall. When the above and following directions have been carried out, and the height of the valve-seat determined, it will be observed that to draw this center-line higher would raise the valve-seat, and that to draw it lower would require that the exhaust passageway, x , be carried farther around the cylinder-wall, thus cooling it and aiding condensation in the cylinder.

68. Next calculate the area of the exhaust-pipe so that the steam velocity in the pipe shall be from $\frac{2}{3}$ to $\frac{3}{4}$ of the velocity through the port. Check-rules for this diameter are: (1) That the area of exhaust-pipe should be 50% greater than the area of the steam-port; (2) That the diameter of the exhaust-pipe should be about $\frac{1}{3}$ the bore of the cylinder. Look up in "Kent," under "Pipes—Wrought Iron," and find the thickness and outside diameter of exhaust-pipe determined upon, and then draw in place, as shown in Fig. 29. In marine work, especially where copper pipes are used on account of their flexibility, the exhaust-pipe boss is flanged, as shown by dotted lines in Fig. 29, and the pipes are flanged correspondingly and bolted fast. For proportions of this flange, number of bolts, etc., see "Kent."

69. The horizontal line through *i* may be drawn about $\frac{1}{16}$ " above the pipe thread so as to allow clearance for pipe tap. Draw the edge line of the port, *k j*, and round off the corner with the arc *j i*. Measure the radial distance, *x*, of the smallest section at this point, and multiply by *y*, its average width as obtained in paragraph 66. This should give an area = $1.33 \times$ steam-port area. If it should be lacking only a small amount, *x* may be increased by using the dotted arc, *t u*, with center at *g*, as the outline of the cylinder. If it is still too small, another trial height of valve-seat must be taken so that lines *a g* and *b h* will spread more and give a greater value to *y*.

70. The steam-chest may be cast solid with the cylinder on smaller engines, or may be, especially in larger ones, cast separately and bolted down. The plan of the steam-chest may be rectangular, as shown in Fig. 30, or it may be circular. In either case the opening under the steam-chest cover must be large to allow putting the valve in place. The steam-chest walls on all sides may be placed as close to the valve-seat center as the valve-travel will permit, and the flange placed entirely on the outside of the wall; or, the walls may be so placed as to have the flange outside on two sides and inside on two sides, as shown in Figs. 28 and 29. Sometimes the flanges are placed on the inside of all four walls. Note that allowances for placing the steam-pipe should be made in planning the walls and flanges of the steam-chest. The valve, when in its extreme position should not be so close to the end of the steam-pipe as to obstruct the flow of steam.

71. The thickness of the steam-chest walls may be made the same as the cylinder-walls, or if the latter have been taken rather heavy the chest-wall may be thinner. The flanges may be 25% greater than the cylinder-wall. The steam-chest cover flange may be made slightly less than the chest-wall flange, and the steam-chest cover equal to or a little less than the chest-wall. The rules for width of flange, and diameter and number of bolts for the steam-chest, are based on the same principles which apply to the cylinder and cylinder-cover flanges. Sometimes the thickness of the steam-chest flange may not figure equal $1\frac{1}{4}$ times stud-bolt diameter, which is necessary to give the latter a proper hold. Then a boss may be cast on the under side of the flange, as shown in dotted lines for the bolt, *o*, in Figs. 29 and 30.

72. The steam-chest cover may have a drop-seat of from $\frac{1}{8}$ " to $\frac{1}{4}$ ", as shown in Figs. 28 and 29. This is desirable in serving as a guide inasmuch as the bolt holes in the steam-chest cover are larger than the bolts. The height of the steam-chest may be such that the distance between the top of the valve and the under side of the steam-chest cover equals, at least, the width of the steam-port.

73. In case the steam-chest wall in Fig. 28 is drawn closer to the valve-seat and the flange placed outside, the faced surface, *p*, for the seat of the screwed valve-stem stuffing-box must be moved out to the end of the flange in order that the stuffing-box nut, *f*, Fig. 31, may be conveniently accessible in screwing on.

74. The diameter of the valve-stem, d , Fig. 31, may be found by taking the total load on the valve $= l \times b \times p$ (l = length, b = breadth of valve, and p = initial steam pressure), and multiplying by a coefficient of friction, which in cases of this kind is taken as high as 20 or 25% due to effort to start valve, especially after a period of idleness. The strain of the valve-stem at the root of the threads should not exceed 2,500 lbs. per square inch for long wrought-iron rods of small diameter and 3,500 lbs. for steel. For engines above, say, an 8" or 9" bore and ordinary length of rod, 3,000 lbs. may be used for wrought iron and 5,000 lbs. for steel. Having calculated the necessary area at

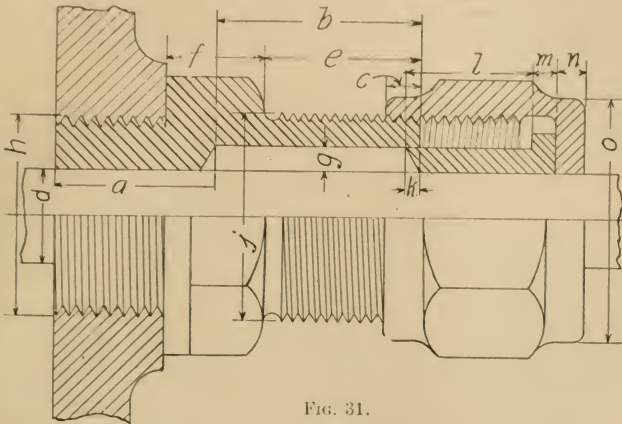


FIG. 31.

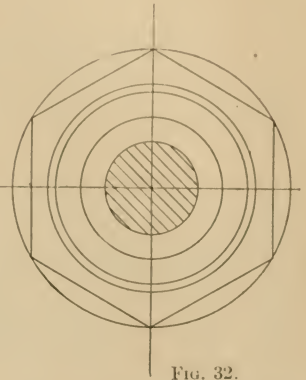


FIG. 32.

the root of the thread, the outside diameter may be readily found in the table on "U. S. or Sellers System of Screw-Threads," in "Kent." In practice the diameter of the valve-rod will be usually found to be $\frac{1}{3}$ to $\frac{1}{2}$ that of the piston-rod if both are made of the same material.

75. The valve-stem stuffing-box is usually made of brass, and screwed in place, see Fig. 31. The following proportions for the box, gland, and bushings may be used:

$a = 1\frac{1}{2} d.$	$b = 2 d + \frac{1}{8}''.$	$c, n = \frac{1}{8} \text{ to } \frac{1}{4} d.$
$e = 1\frac{3}{8} d.$	$f = d + \frac{1}{16}''.$	$g = \frac{1}{4} d + \frac{1}{32}''.$
$h = 2 d + \frac{1}{8}''.$	$j = 2 d + \frac{1}{8}''.$	$k = \frac{1}{8} d.$
$l = 1\frac{1}{8} d.$	$m = \frac{1}{4} d.$	$o = 2.5 d + \frac{3}{16}''.$

These proportions give a *long* stuffing-box, which has the advantage that the packing does not require to be very tightly compressed. When circumstances require it, this length may be reduced.

76. The type of stuffing-box shown in Fig. 31, although used principally on the valve-rod in engine work, is sometimes used also for the piston-rod. Instead of the hexagonal form the gland is sometimes made circular with longitudinal grooves, or sometimes with radial holes, in either case a corresponding form of spanner being used to screw up the gland.

77. The steam pipe may be placed as shown in Fig. 28 at A , or in large engines the steam-chest may be cast separate and bolted to the engine cylinder. The diameter of the pipe may be calculated, using a steam velocity at least equal to that used in computing the steam-port. A rough rule in practice is that the steam-pipe should be about $\frac{1}{4}$ the cylinder-bore.

78. In finishing the design, place on the necessary working dimensions and indicate the finished surfaces.



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